(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 937 894 A2

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
25.08.1999 Bulletin 1999/34

(51) Int. Cl.^{6}: F04C 18/16, F04C 29/00, F04C 29/10

(21) Application number: 99108729.7

(22) Date of filing: 18.08.1995

(84) Designated Contracting States:
DE FR GB

(30) Priority: 19.08.1994 JP 21816394

(62) Document number(s) of the earlier application(s) in accordance with Art. 76 EPC:
95305786.6 / 0 697 523

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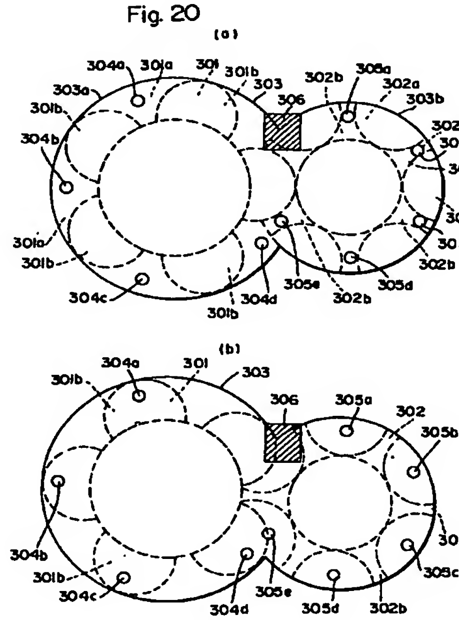
Remarks:
This application was filed on 03 - 05 - 1999 as a divisional application to the application mentioned under INID code 62.

(54) Screw fluid machine and screw gear used in the same

(57) A screw fluid machine is disclosed particularly for use as a vacuum pump. The machine consists of a male rotor (301) having teeth mated with a female rotor (302) and mounted for rotation in a casing (109) to form fluid chambers. Discharge ports (304,305) are formed in a screw end face plate (303a,303b) constituting a part of the casing (109) so that an end face of the teeth of each rotor (301,302) opens and closes each discharge port (304,305). A set of discharge ports (304) are provided on the male rotor side whose number is smaller than the total number of teeth of said male rotor (301) and a set of discharge ports (305) are provided on the female rotor side whose number is smaller than the total number of teeth of said female rotor (302). Each of the discharge ports (304,305) are provided with a pressure adjustment device (307) which opens when the pressure in said chambers exceeds atmospheric pressure.

Fig. 20



Description

[0001] This application is a division of EP-A-0697523 application No. 95305786.6.

[0002] The present invention relates to a screw fluid machine such as a screw-type pump, a screw-type compression pump, a screw-type motor or the like, and particularly to a screw vacuum pump which is suitably used in a low/medium vacuum range from the atmospheric pressure to 10^{-4} Torr level in vacuum degree, and also relates to a screw gear which is suitably used for the screw pump or the like.

[0003] Various types of vacuum pumps such as an oil-sealed rotary vacuum pump, a Roots pump, a diffusion pump, etc. have been hitherto used in a low/middle vacuum range.

[0004] For example, in a manufacturing field for semiconductors, wafers are subjected to a predetermined treatment while placed in a chamber which is kept in a vacuum state. In this treatment, the chamber is evacuated by a vacuum pump while supplied with inert gas such as N_2 gas or the like to remove impurities (0_2, CO_2, etc.) in the chamber, and finally the chamber is kept in a vacuum state from several Torr to 10^{-4} Torr level. An oil-sealed rotary vacuum pump, a Roots type mechanical booster pump or the like has been utilised as a vacuum pump used in the above semiconductor manufacturing process.

[0005] However, the oil-sealed rotary vacuum pump has a disadvantage that lubricant oil used in this pump is liable to be contacted with various kinds of gas (for example, arsenic, gallium, chlorine, Poly-Si, fluorine, etc.) which are used as reaction gas in the semiconductor manufacturing process, resulting in reduction of the lifetime of the lubricant oil. In addition, it has another disadvantage that a semiconductor manufacturing chamber is contaminated by oil molecules, and this contamination adversely affects the semiconductor manufacturing process.

[0006] Furthermore, this type of pump has a narrower pressure range in which it can work normally, and thus several kinds of pumps must be successively used while changed to another until a desired pressure (vacuum state) is obtained. Therefore, it cannot be performed using only one vacuum pump to evacuate the chamber from the atmospheric pressure to 10^{-4} Torr level.

[0007] In order to solve the above problem, an oil-free screw vacuum pump as disclosed in Japanese Laid-open Patent Application No. Sho-60-216089 has been proposed.

[0008] This type screw vacuum pump as disclosed in the above publication is of an oil-free type, and it can cover the above pressure range using only one pump.

[0009] The screw type vacuum pump as described above will be briefly described hereunder with reference to Figs. 1 and 2.

[0010] Fig. 1 is a cross-sectional view showing a screw-type vacuum pump which corresponds to a plan view, and Fig. 2 is a cross sectional view showing the screw-type vacuum pump of Fig. 1 which corresponds to a side view. As shown in Figs. 1 and 2, a male rotor 10 and a female rotor 11 are freely rotatably supported through bearings 14, 15, 16 and 17 in a main casing 12 and a suck-in casing 13, and each of the male rotor 10 and the female rotor 11 comprises a screw gear (screw). The screw gear has a fixed helix angle of tooth trace at all times, and further it has a fixed tooth-trace pitch in its rotation-axis direction (hereinafter referred to as "tooth pitch of rotational axis") and a fixed toot-trace pitch on the plane of rotation which is vertical to the rotation axis (hereinafter referred to as "tooth pitch of rotational plane"). Therefore, these pitches are not varied in accordance with variation of the rotational angle of the rotors 10 and 11.

[0011] In Figs. 1 and 2, a suck-in side 10a of the rotors is kept at a low pressure of 10^{-4} Torr level while a discharge side 10b of the rotors is kept at the atmospheric pressure, so that a radial load imposed on the rotors is extremely smaller at the suck-in side than the discharge side. Therefore, the bearings 14 and 15 of the suck-in side are designed to support a radial load and a thrust load with deep groove ball bearings, and the bearings 16 and 17 at the discharge side are designed to support only a radial load with cylindrical roller bearings.

[0012] Timing gears 18 and 19 are secured to the shaft ends of the rotors 10 and 11 to adjust the gap interval between the male and female rotors 10 and 11 so that these rotors do not come into contact with each other.

[0013] Lubrication of the bearings 14 and 15 is performed by oil splash. That is, lubricant 21 stocked in a suck-in cover 20 is splashed to the bearings 14 and 15 by the timing gears 18 and 19. Likewise, lubrication of the bearings 16 and 17 is also performed by a disc 22 which is secured to the shaft of the male rotor. That is, lubricant 24 stocked in a discharge cover 23 is splashed to the bearings 16 and 17 by the disc 22. Furthermore, shaft seals 25, 26, 27 and 28 are provided to prevent leakage of the lubricant of the bearings and timing gears into chambers.

[0014] Since substantially the atmospheric pressure is kept in a chamber 10b at the discharge side of the rotors and in the discharge cover 23, 50 that the differential pressure acting on the shaft seals 27 and 28 at the discharge side is relatively small. On the other hand, since a chamber at the suck-in side is kept at a pressure of 10^{-4} Torr level, the differential pressure acting on the suck-in side shaft seals 25 and 26 becomes large when the inside of the suck-in cover 20 is released to the atmospheric air, so that it is difficult to keeping a seal effect at the suck-in side. Accordingly, in order to enhance the sealing effect, the inside of the suck-in cover 20 is designed to intercommunicate with a low-pressure chamber 10C through exhausting pipes 29 and 30 to reduce the pressure in the suck-in cover 20 and thus reduce the

ment between the timing gears, so that a working environment becomes worse.

[0027] --Still-furthermore, in another conventional screw vacuum pump, pressure adjustment devices 50 as shown in Fig. 4 are provided on the lower surface of the casing 12 and in the axial direction of the rotors in order to prevent excessive rise-up of the pressure of the chambers and thus prevent the abnormal temperature rise-up of the vacuum pump when the vacuum pump works in a state where the suck-in pressure is substantially equal to the atmospheric pressure.

[0028] As shown in Fig. 5, the pressure adjustment device includes a discharge port 52 provided to the lower portion of the casing 12, a valve rod 53 for opening and closing the discharge port 52, a spring 54 for supporting the dead weight of the valve rod 53, a valve box 55 for accommodating the valve rod 53 and the spring 54, and an air open port 56 for discharging to the outside the gas discharged from the discharge port 52 which is formed in the valve box 55. An 0-ring is secured around the valve rod 53. When the pressure adjustment device 50 as shown Fig. 5 is disposed as shown in Fig. 4, in some cases a chamber 51a and a chamber 51b intercommunicate with each other through the discharge port 52 as shown in Fig. 5, and the gas flows from the chamber 51a to the chamber 51b in a direction as indicated by an arrow. That is, each addendum 58 of the rotors does not have sufficient width, so that there occurs a case where the discharge port 52 is located over both the neighbouring chambers 51a and 51b. As a result, the gas leaks from the high-pressure chamber 51a to the low-pressure chamber 51b, and thus it takes a long time to evacuate the suck-in side to a desired vacuum degree.

[0029] An object of the present invention is to provide a screw vacuum pump in which increase in shaft torque due to excessive compression can be prevented, abnormal rise-up of temperature can be prevented and the pressure at the suck-in side can be reduced to a desired vacuum degree for a short time.

[0030] In order to attain the object of the present invention there is provided a screw fluid machine in accordance with claim 1.

[0031] In the screw fluid machine thus constructed, the discharge valve of the pressure adjustment device closes the outside of the discharge port when the suck-in pressure is low and the pressure in the chambers is lower than the atmospheric pressure or its peripheral value.

[0032] At this time, the inside of the discharge port is closed by the tooth end face of the screw gear constituting the rotor, and thus a chamber does not intercommunicate with an adjacent chamber even when the rotors are rotated, so that the gas leakage from a high-pressure chamber side to a low-pressure chamber side can be prevented and thus the pressure at the suck-in side can be evacuated to a desired vacuum degree for a short time.

[0033] In addition, when the pressure of the suck-in gas is higher and the pressure in the chambers is higher than the atmospheric pressure or its peripheral value, the discharge valve of the pressure adjustment device is released, and the gas in the chambers is discharged from the discharge port to the outside. Furthermore, when the suck-in pressure is reduced and the pressure in the chambers does not reach the atmospheric pressure just before the chamber intercommunicates with the discharge port, all the discharge ports of the pressure adjustment device are closed, and the gas in the chambers is discharged from the discharge port under pressure without being discharged from the pressure adjustment device to the outside.

Fig. 1 is a cross-sectional view showing a conventional screw vacuum pump, which is taken along a line B-B of Fig. 2;

Fig. 2 is a cross-sectional view showing the conventional screw vacuum pump of Fig. 1, which is taken along a line A-A of Fig. 1;

Fig. 3 is a schematic diagram showing an engagement state of male and female rotors of the conventional screw vacuum pump which is developed in a peripheral direction of the rotors; Fig. 4 is a cross-sectional view showing the conventional screw vacuum pump;

Fig. 5 is a cross-sectional view showing a main part of a pressure adjustment device shown in Fig. 4;

Fig. 6 is a plan view of a screw gear used in the present invention;

Fig. 7 is a development on an engagement pitch cylinder of the screw gear used in the present invention, which shows a tooth-trace rolling curve of a parabola (quadratic curve) on the coordinates in which the abscissa represents the male rolling peripheral length of the engagement pitch cylinder and the ordinate represents a helix advance amount;

Fig. 8 is a diagram showing the rise-up of the temperature of the screw vacuum pump of the present invention and the conventional screw vacuum pump, in which a dotted line represents the conventional screw vacuum pump and a solid line represents the screw vacuum pump of a first embodiment of the present invention;

Fig. 9 is a perspective view showing male and female rotors which are used in the first embodiment of the present invention;

Fig. 10 is a plan view showing the male and female rotors of Fig. 9;

Fig. 11 is a cross-sectional view showing the screw vacuum pump in which the male and female rotors shown in Figs. 9 and 10 are used;

Fig. 12 is a cross-sectional view of the screw vacuum pump which is taken along a line A-A of Fig. 11;

male and female rotors extends (starts) from the point (origin) at which the male and female rotors are contacted and coincident with each other on the pitch cylinder (that is, X_M=0 and x_F=0), and on both the curves, y increases as x increases. That is, for the male rotor, y is a monotonically increasing function of x_M, and for the female rotor, y is a monotonically increasing function of x_F

[0045] This is equivalent to such a condition that x and y are interchanged with each other to regard y as an independent variable and regard x as a function of y. That is, for the male rotor, x_M is regarded as a monotonically increasing function of y and represented as follows:

$$x_M = F_M(y) \tag{1}$$

For the female rotor, x_F is regarded as a monotonically increasing function of y and represented as follows:

$$x_F = F_F(y) \tag{2}$$

[0046] Furthermore, since both the curves pass through the origin,

$$F_M(O) = F_F(O) = 0 \tag{3}$$

Here, in the following equations, parameters

$$\beta_{Mg},\quad \beta_{Fg},$$

θ_M and θ_F which are defined as follows are introduced:

β_{Mg}: helix angle of the male rotor on the pitchcylinder

β_{Fg}: helix angle of the female rotor on the pitch cylinder

θ_M : rotational angle of the male rotor

θ_F : rotational angle of the female rotor

The helix angles

$$\beta_{Mg},\quad \beta_{Fg}$$

corresponds to the angles shown in Fig. 7.

Furthermore, representing the radius of the pitch cylinder of the male (female) rotor by R_M (R_F), the rotational angles θ_M, θ_F are represented as follows:

$$\theta_M = x_M/R_M \tag{4}$$

$$\theta_F = x_F/R_F \tag{5}$$

[0047] Using the equations (1), (2), the helix angles

$$\beta_{Mg},$$

The development of a tooth-trace rolling curve of rotors having another tooth shape is obtained by parallel shifting x=F(y) in the x-axis direction by mT. Here, m represents a positive or negative integer. These curves are represented by dotted lines in Fig. 7.

[0052] As the simplest example, the following quadratic function can be selected as F(y):

$$F(y) = Ay^2 + By \ (A>0,\ B>0) \qquad (17)$$

The curve shown in Fig. 7 is an example of the quadratic curve as described above.

[0053] With respect to the helix-angle variable type screw gear thus specified, the development of the tooth-trace rolling curve on the pitch cylinder is given as any function satisfying the equation (14). Therefore, on the basis of variation of the gradient of the curve, the tooth-trace helix angle on the pitch cylinder is varied in accordance with the rotational angle of the screw, and further on the basis of the variation of the gradient of the curve, the tooth-shaped portion is determined in consideration of the basic technical idea of the tooth-trace helix angle of an existing helical gear or screw gear. The planeof-rotation pitch T is made coincident on the pitch cylinders to perform an engagement, and the helix is advanced in the rotational-axis direction (y-direction) while the pitch ta of the rotational axis direction varies momentarily with variation of the rotational angle, but the engagement state and the toothshape status on the plane of rotation are kept.

[0054] That is, the rolling peripheral length and the helix advance direction amount on the pitch cylinders are equal between the male and female rotors, so that the length of the helix on each pitch cylinder is equal between the male and female rotors. That is, in any variable range of y [yi, yj],

$$\int_{yi}^{yj} (dx_F^2 + dy^2)^{1/2} = \int_{yi}^{yj} (dx_M^2 + dy^2)^{1/2} \quad \ldots\ldots \quad (A)$$

From the equation (A), the length of the helix on each pitch cylinder in the variable range [yi,yj] is equal between the male and female screws to perform the engagement of both the screws.

[0055] Furthermore, the tooth-trace rolling curve is also expressed by a function of the rotational angle, and the rotational angle and the tooth-trace rolling amount are proportional to each other. The length of the helix at the diameters R_M' and R_F' other than the pitch diameters of the male and female tooth-shaped portions can be obtained by replacing the x_M and x_F in the equation (A) with the following equations using the equations (4) and (5):

$$x'_M = x_M R_M'/R_M \quad x'_F = x_F R_F'/R_F$$

Accordingly, the equation (A) is not satisfied at the contact portion of the diameter other than that of the pitch cylinder, and it is adjusted by slip. That is, the following equation is satisfied:

$$\int_{yi}^{yj} (dx'_F{}^2 + dy^2)^{1/2} + (\text{slip amount}) = \int_{yi}^{yj} (dx'_M{}^2 + dy^2)^{1/2} \quad \ldots\ldots \quad (A)$$

[0056] In order to enable the engagement between the male and female rotors, the following relationship must be satisfied between the rotational angles θ_M and θ_F:

$$\theta_M N_F = \theta_F N_M \qquad (18)$$

Here, N_M and N_F represent the number of teeth of the male and female rotors, respectively. Furthermore, the radius R_M,

the rotational axis.

[0060] That is, the screw thus constructed has not only characteristics as an ordinary screw gear, but also characteristics as a screw having high sealing property on the plane of rotation. In addition, the rotation-axis pitch can be varied periodically and continuously.

[0061] Accordingly, when the male and female rotors are designed using this screw gear, the tooth-trace helix angles of the male and female rotors vary in accordance with the rotational angle of the rotors, so that the volume of the V-shaped chambers formed by the rotors and the casing can be continuously varied. That is, all the chambers can be designed so that the volume thereof is reduced.

[0062] As described above, when a screw vacuum pump or a compression pump is constructed with the screw gear as described above, the volume of the chambers varies continuously to perform a continuous compression and feeding action, so that the temperature of the pump gradually increases from the suck-in side to the discharge side, as indicated by a solid line of Fig.8, and there occurs no local rise-up in temperature.

[0063] Furthermore, each chamber has a suck-in action for sucking gas into the chamber in a state where it intercommunicates with the induction port, a continuous gas compressing and feeding action for continuously compressing and feeding the gas in the chamber, and a discharge action for discharging the gas to the outside in a state where it intercommunicates with the discharge port (that is, it has no mere feeding action), so that the screw vacuum pump can be effectively operated.

[0064] Still furthermore, since the rotation-axis pitch is variable, the total length of the rotors can be more shortened as compared with the conventional screw fluid machine using the fixed rotation-axis pitch, so that the screw fluid machine can be designed in a compact size.

[0065] Next, another embodiment in which a Roots portion is provided at least one end side of each screw portion of the male and female rotors in the screw fluid machine of the present invention will be described with reference to Figs. 9 to 12.

[0066] Fig. 9 is a perspective view showing male and female rotors used in this embodiment, and Fig. 10 is a plan view showing the male and female rotors of Fig. 9. Fig. 11 is a cross-sectional view showing a screw vacuum pump using the male and female rotors shown in Fig. 10, and Fig. 12 is a cross-sectional view of the screw vacuum pump of Fig.11 which is taken along a line A-A of Fig. 11..

[0067] As described above, each of the conventional male and female rotors is provided with a single screw gear. On the other hand, this embodiment is characterised in that each of the male and female rotors is provided with the screw gear as described above and a Roots.

[0068] As shown in Figs. 9 and 10, a male (female) rotor 101 (102) comprises a screw gear portion 101a (102a), and male-side Roots portions 103 and 105 (female-side Roots portions 104 and 106). The male-side Roots portions 103 and 105 (female-side Roots portions 104 and 106) are formed at both ends of the screw gear portion 101a (102a).

[0069] Chambers 101b (102b) which are formed by the screw gear portion 101a (102a) of the male (female) rotor 101 (102) and the casing intercommunicate with chambers 103a (104a) which are formed by the male-side Roots portion 103 (female-side Roots portion 104) and the casing, and likewise the chambers 101b (102b) intercommunicate with the chambers 105a (106a) which are formed by the male-side Roots portion 105 (female-side Roots portion 106) and the casing. A rotational shaft 107 (108) is formed at one end portion of the male (female) rotor 101 (102).

[0070] Next, an arrangement state of the male and female rotors 101 and 102 in the casing will be described with reference to Figs. 11 and 12.

[0071] As shown in Figs. 9, 10, 11, 12 the male rotor 101 and the female rotor 102 are accommodated in a main casing 109, and these rotors are freely rotatably supported through bearings 111 and 112 which are secured to an end plate 110 for sealing one end surface of the main casing 109, and bearings 118 and 119 which are secured to an auxiliary casing 117.

[0072] A discharge port 109b for discharging to the outside gas which are compressed by the male and female rotors 101 and 102 is provided at the end plate 110 side of the main casing 109. Furthermore, seal members 113 and 114 are secured to each of the bearings 111 and 112, and these seal members 113 and 114 are used to prevent lubricant oil from invading into the chambers from timing gears 115 and 116 as described later.

[0073] The timing gears 115 and 116 which are accommodated in the auxiliary casing 117 are secured to the rotational shafts 107 and 108 of the male and female rotors 101 and 102 to adjust the gap interval between the male and female rotors so that these rotors are not contacted with each other.

[0074] The bearings 111 and 112 are lubricated by oil splash, that is, lubricant oil (not shown) stocked in the auxiliary casing 117 is splashed to the bearings 111 and 112 by the timing gears 115 and 116. The auxiliary casing 117 is secured to the other end of the main casing 109, and a induction port 109a is secured to the other end side of the main casing 109.

[0075] In the screw vacuum pump thus constructed, as shown Fig. 9, 10, through rotation of the male and female rotors 101 and 102, gas is sucked from the induction port 109a into the chambers 103a and 104a which are formed by the male-side Roots portion 103, the female-side Roots portion 104 and the casing. At the suck-in time, the sucked gas

[0088] Like the embodiment of Fig. 16, the motors M^1 and M^2 are connected to the inverters 202 and 203 for transmitting the driving alternating signal or driving pulse signal, and the inverters 202 sand 203 are connected to the controller 204 for transmitting a control signal to control the frequency of the inverters 202 and 203. This control system is further provided with feedback circuits 205 and 206 which receive the driving alternating signals or driving pulse signals from the inverters 202 and 203 respectively. Each of the feedback circuits 205 and 206 transmit a control signal to each of the inverters 202 and 203.

[0089] When a control signal corresponding to a prescribed rotational number is transmitted from the controller 204 to the inverters 202 and 203, a driving alternating signal or driving pulse signal having a prescribed frequency (reference frequency) is transmitted from each of the inverters 202 and 203 to each of the motors M^1 and M^2.

[0090] Here, if the driving alternating signal or driving pulse signal transmitted from each of the inverters 202 and 203 is deviated from the reference frequency due to a frequency error of the inverters 202 and 203 or the like, the male and female rotors 101 and 102 cannot be rotated in synchronism with each other. However, the driving alternating signal or driving pulse signal transmitted from each of the inverters 202 and 203 is input to each of the feedback circuits 205 and 206. Each of the feedback circuits 205 and 206 serves to correct the frequency error of each of the inverters 202 and 203, and supplies each of the inverters 202 and 203 with such a control signal that the frequency of each inverter 202, 203 is coincident with the reference frequency. As a result, the driving alternating signal or driving pulse signal which is transmitted from each of the inverters 202 and 203 gradually approaches to the reference frequency, and finally the male and the female rotors 101 and 102 are rotated in synchronism with each other.

[0091] As described above, even if there is any frequency error between the inverters 202 and 203, the feedback circuits 205 and 206 work to transmit the control signals from the feedback circuits to the inverters 202 and 203 so that the error is reduced. Therefore, the rotation of the male rotor 101 and the rotation of the female rotor 102 is synchronised with each other, so that the load applied to the timing gears 115 and 116 is gradually reduced and thus the noise due to the engagement of the timing gears can be suppressed.

[0092] In the above embodiment, the helix angle of the screw gear may be set to continuously vary or not to continuously vary, and furthermore, the Roots portion may be provided to the rotors.

[0093] Figs. 18 and 19 are diagrams showing a improved modification of the vacuum pump shown in Figs. 14 and 15. The vacuum pump of this modification is provided with Roots portions 213 and 214, screw portions 215 and 216, Roots portions 217 and 218, screw portions 219 and 220 and Roots portions 221 and 222 in this order from the left side to the right side in the rotational axial direction. The motors M^1 and M^2 which are controlled in the same manner as described above are secured to one end sides of rotational shafts 223 and 224, respectively.

[0094] By this arrangement of the motors M^1 and M^2, the motors M^1 and M^2 can be easily secured to the rotational shafts 223 and 224 even when the motors M^1 and M^2 have a large diameter. The respective parts of right and left screws 215, 218, 219 and 220 which are provided on the same axial line are designed to have opposite helixes so that the gas sucked from the induction port 225 is branched into two parts in the right and left directions and then discharged from the discharge ports 226 and 227, respectively. The other construction is similar to that of Figs. 14 and 15. Accordingly, the same elements as Figs. 14 and 15 are represented by the same reference numerals, and the description thereof is omitted.

[0095] Next, an embodiment in which a pressure adjusting valve is provided to the vacuum pump of the present invention will be described with reference to Figs. 20 to 22.

[0096] Fig. 20 is a schematic diagram showing a discharge-side end face plate portion (inner wall surface portion) of the casing of the screw vacuum pump, which is viewed from the rotor side. In Fig. 20, (a) shows a state where the tooth end surface of the male rotor is not located at the discharge port of the male rotor side, and (b) shows a state where the tooth end surface of the male rotor is located at the discharge port because the male rotor is rotated. Fig. 21 is a schematic diagram of the screw vacuum pump which is developed in the peripheral direction of the rotors, and Fig. 22 is an enlarged view showing a main portion of the discharge port.

[0097] As shown in these figures, a male rotor 301 and a female rotor 302 are accommodated in a casing 303 like the conventional screw vacuum pump.

[0098] A male rotor end face plate 303a and a female rotor end face plate 303b (in Fig. 21) are formed at the discharge side of the casing 303. The end face plate 303a and the end face plate 303b are not contacted with the tooth end face of the male rotor 301 and the tooth end face of the female rotor 302, and these plates are disposed away from these rotors at minute gap intervals. Accordingly, the gas tightness of chambers 301a and 302a are kept by the male and female rotor end face plates 303a and 303b and the tooth end faces 301b and 302b of the male and female rotors 301 and 302.

[0099] Furthermore, discharge ports 304a, 304b, 304c and 304d are formed on the end face plate 303a of the male rotor 301, and also discharge ports 305a, 305b, 305c, 305d, 305e are formed on the end face plate 303b of the female rotor. In addition, a discharge port 306 is formed at the upper portions of the end face plate 303a and the end face plate 303b while extend over these end face plates 303a and 303b.

[0100] There are provided four discharge ports 304 on the male rotor side end face plate 303a, whose number is

port (304, 305) through the gap in the valve box 55 and the air open port 56 to the outside.

[0113] Thereafter, when the suck-in pressure is lowered and the pressure in the chamber concerned does not reach the atmospheric pressure just before the chamber intercommunicates with the discharge port, all the discharge ports 304 and 305 of the pressure adjusting devices are closed, and the gas in the chamber is discharged from the discharge port 306 under pressure without being discharged from the pressure adjusting device 307 to the outside.

[0114] As described above, according to the screw vacuum pump of this embodiment, through the rotation of the rotors of the screw vacuum pump, the insides of the discharge ports are closed by the end tooth faces of the rotors in a state where the tooth end faces of the rotors are located at the discharge ports. Therefore, a chamber can be prevented from intercommunicating with an adjacent chamber through the discharge ports, and no gas leaks from a high-pressure chamber to a low-pressure chamber, so that it does not take a long time to evacuate the suck-in side at a desired vacuum degree.

[0115] Furthermore, the pressure in the chambers are suppressed to a value below the atmospheric pressure at all times, so that excessive compression is not carried out even when the vacuum pump is operated in a state where the suck-in pressure is substantially equal to the atmospheric pressure. Therefore, increase of shaft torque can be prevented, and thus power consumption can be suppressed.

[0116] In addition, since excessive compression is not carried out, the temperature of the screw vacuum pump can be prevented from rising up abnormally, and the dimensional precision of the engagement between the casing and the rotors and the engagement between the male and female rotors, etc. can be kept excellent.

[0117] In the above embodiments, the screw vacuum pump is provided with the four or five discharge ports. However, the number of the discharge ports is not limited to a specific one, and it may be suitably selected in consideration of its use range, its performance, etc.

[0118] Furthermore, the discharge ports are located at the position corresponding to the pitch circle of the screw gear of the rotor. However, the location position of the discharge ports is not limited to this position, and these may be located at such a position that these discharge ports can be closed by the tooth end face of the screw gear.

[0119] In the above embodiments, the urging force of the spring is set to the extent that the dead weight of the valve rod 53 can be supported by the spring. However, it is not limited to this degree, and it may be altered in consideration of the use range, performance, etc. of the screw vacuum pump.

[0120] Furthermore, in the above embodiments, the helix angle of the screw gear may be continuously altered or not continuously altered. In addition, the Roots portion may be provided at the discharge side of the screw portion of the rotor as shown in Figs. 11 and 12 (the discharge-side end face corresponds to the tooth end face).

[0121] As is apparent from the forgoing, according to the screw fluid machine, the tooth-trace helix angle of each of the male and female rotors is designed to vary in its helix direction. Therefore, the volume of each of the V-shaped fluid chambers which are formed by the rotors and the casing can be continuously increased or decreased in accordance with the rotational angle of the rotors. As a result, the abnormal local rise-up of the temperature can be suppressed, so that the dimensional precision of the engagement between the casing and the rotors and the engagement between the male and female rotors can be improved.

[0122] Furthermore, the following screw gear is usable for the screw fluid machine according to the present invention. That is, the screw gear of this invention is characterised in that the peripheral length of the pitch cylinder in the helix advance direction on the development of the tooth-trace rolling curve on the pitch cylinder of the screw gear can be expressed by a substantially monotonically increasing function. With this screw gear, the sealing property in the plane-of-rotation direction can be improved, and thus the gas tightness of the fluid chambers can be improved.

[0123] In addition, the screw gear thus constructed can be used as an ordinary transmission gear, and in addition it can effectively treat any load which is varied in the axis direction with time variation because the helix angle is varied with time variation through rotation.

[0124] According to the fluid machine of the present invention, the Roots portion is provided to at least one end side of the screw portion of the male and female rotors. Therefore, when the fluid machine is used as a vacuum pump, the pumping speed can be greatly improved, and the evacuation operation from the atmospheric pressure to the medium vacuum area of 10^{-4} Torr level can be effectively performed using only one vacuum pump at a stable pumping speed. In addition, when the fluid machine of the present invention is used as a compression pump, a high discharge pressure can be obtained.

[0125] Furthermore, according to the fluid machine of the present invention, the male and female rotors are rotated in synchronism with each other. Therefore, even when the rotors are rotated at a high speed, the noise occurring through the engagement of the timing gears can be suppressed.

[0126] Still furthermore, according to the fluid machine of the present invention, through the rotation of the rotors, the insides of the discharge ports are closed by the tooth end faces of the rotors in the state where the tooth end faces of the rotors are located at the discharge ports. Therefore, a chamber can be prevented from intercommunicating with another adjacent chamber through the discharge ports. As a result, gas can be prevented from leaking from a high-pressure working room to a low-pressure chamber, and no surplus (long) time is needed until the suck-in side is evac-

Fig. 1
PRIOR ART

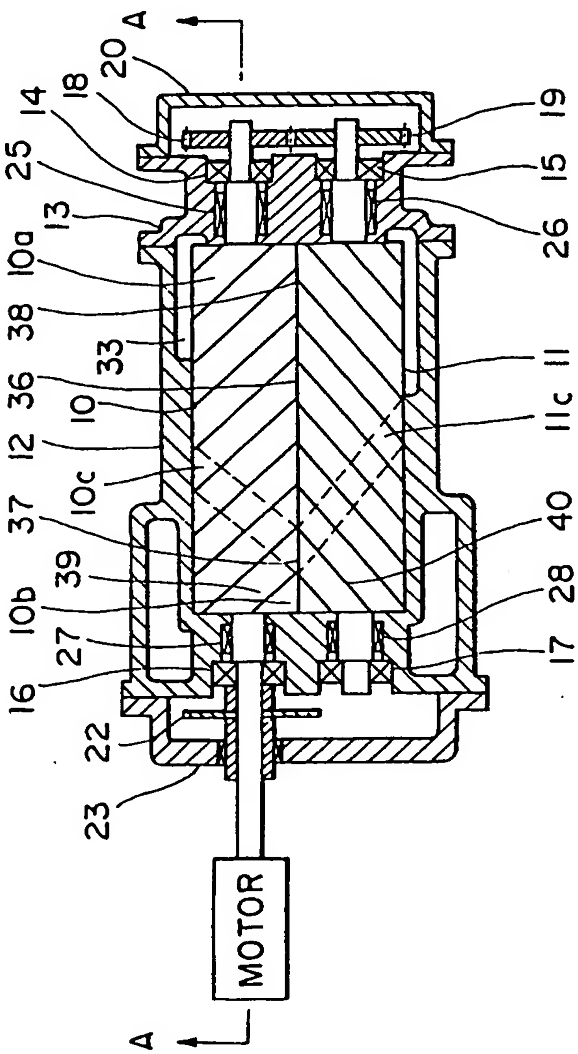


Fig.3 PRIOR ART

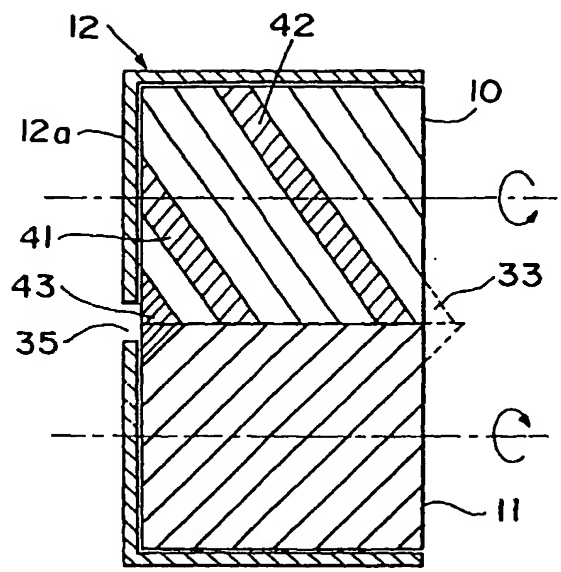


Fig.4 PRIOR ART

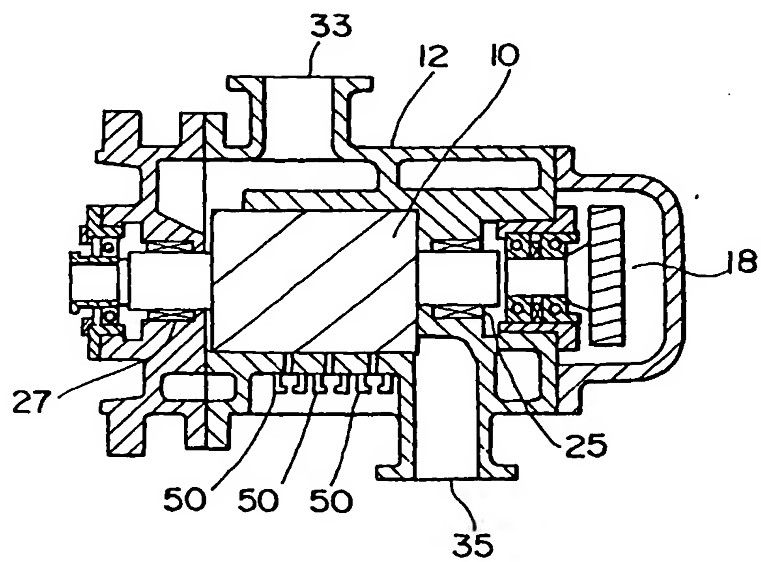


Fig. 7

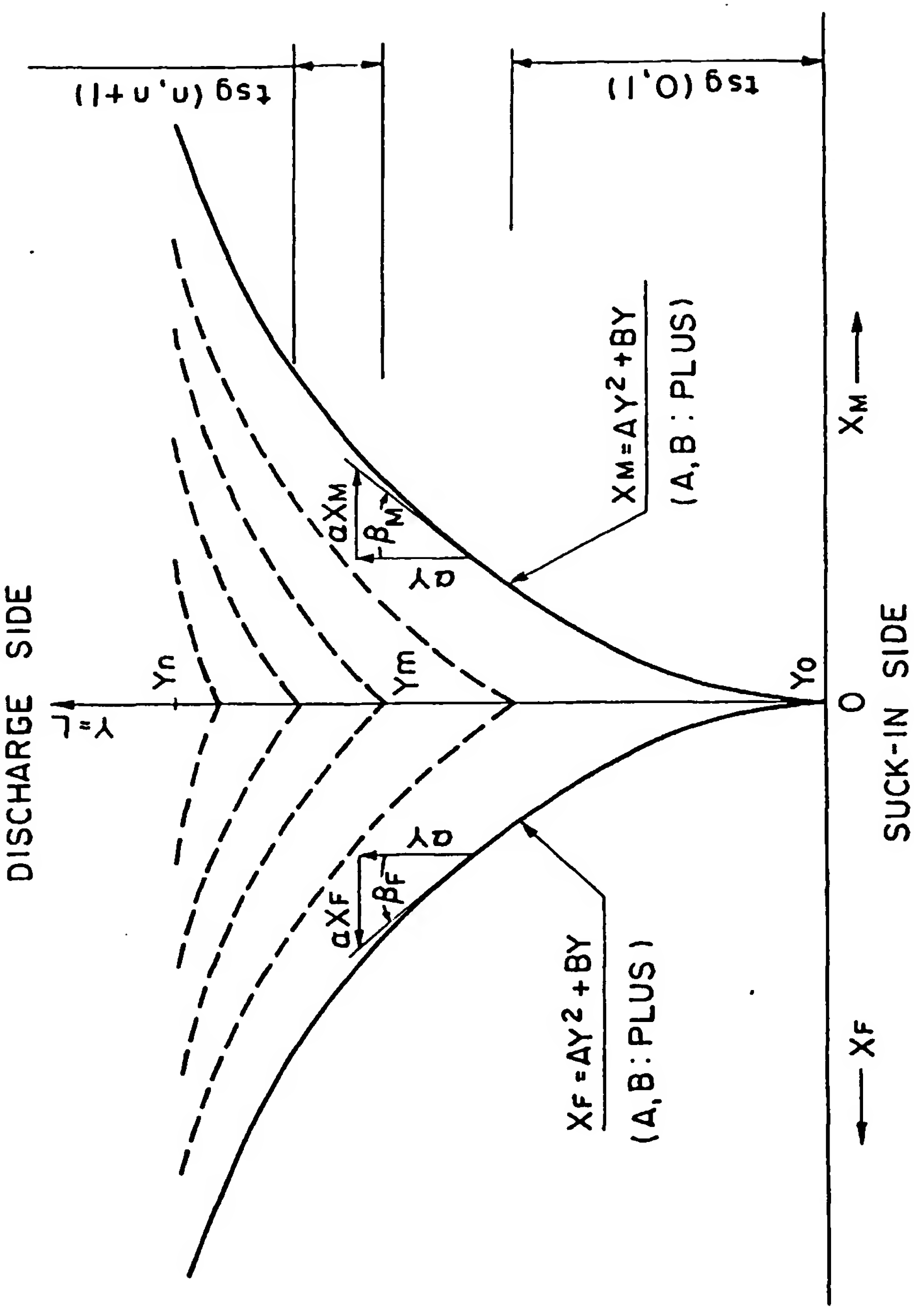

DISCHARGE SIDE
SUCK-IN SIDE
Y=L
Yn
Ym
Yo
O
tsg (n,n+1)
tsg (0,1)
XM=AY2+BY
(A,B : PLUS)
XF=AY2+BY
(A,B : PLUS)
αXM
βM
αY
αXF
βF
αY
XM
XF

Fig. 9

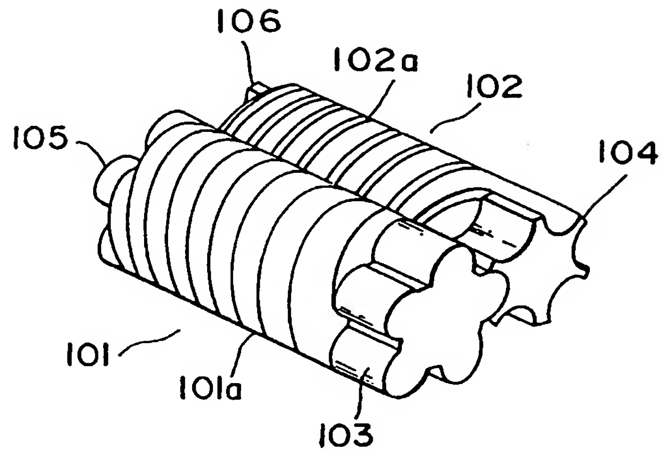

106
102a
102
105
104
101
101a
103

Fig. 10

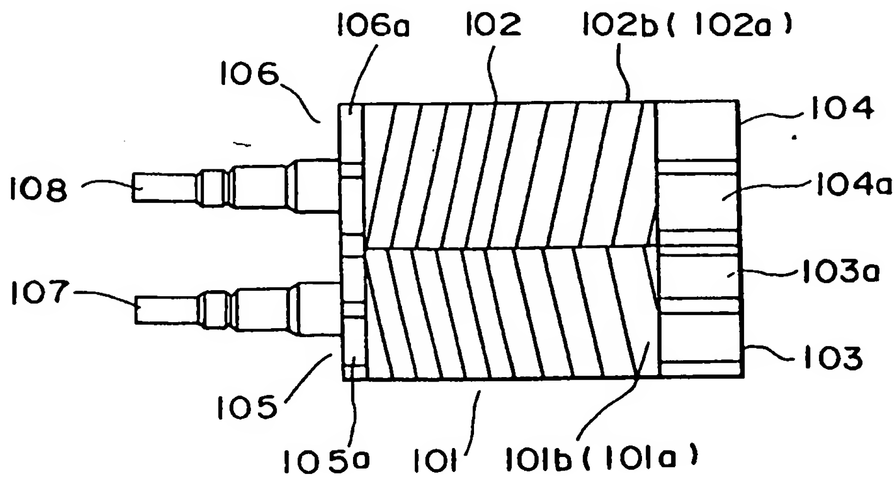

106a
102
102b (102a)
106
104
108
104a
107
103a
103
105
105a
101
101b (101a)

Fig. 13

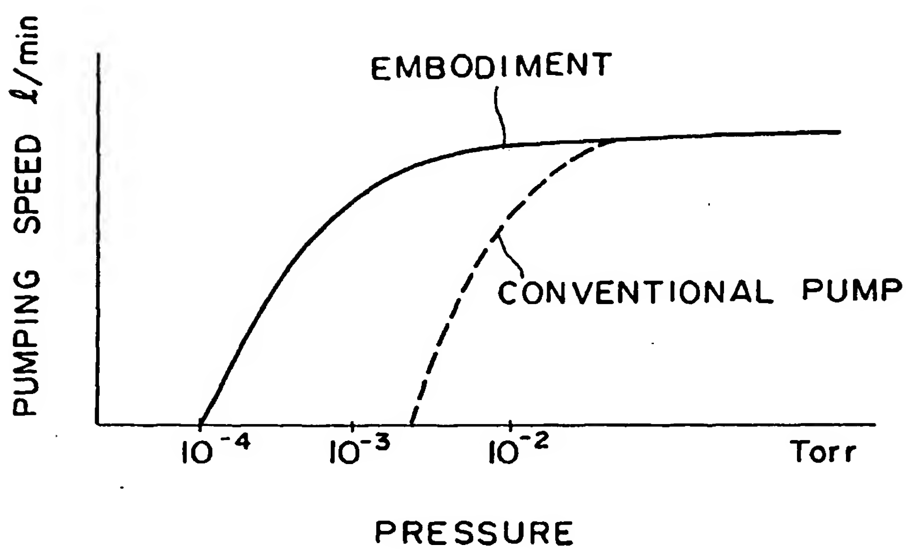

EMBODIMENT
CONVENTIONAL PUMP
PUMPING SPEED ℓ/min
10^{-4}
10^{-3}
10^{-2}
Torr
PRESSURE

Fig. 16

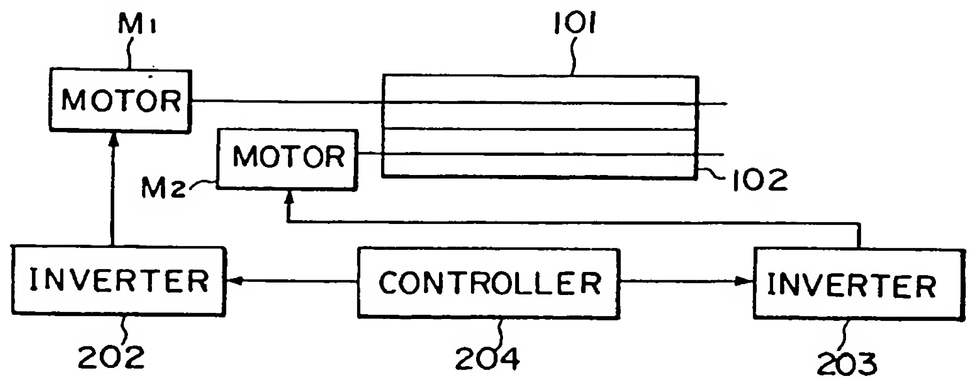

M1
101
MOTOR
MOTOR
M2
102
INVERTER
CONTROLLER
INVERTER
202
204
203

Fig. 17

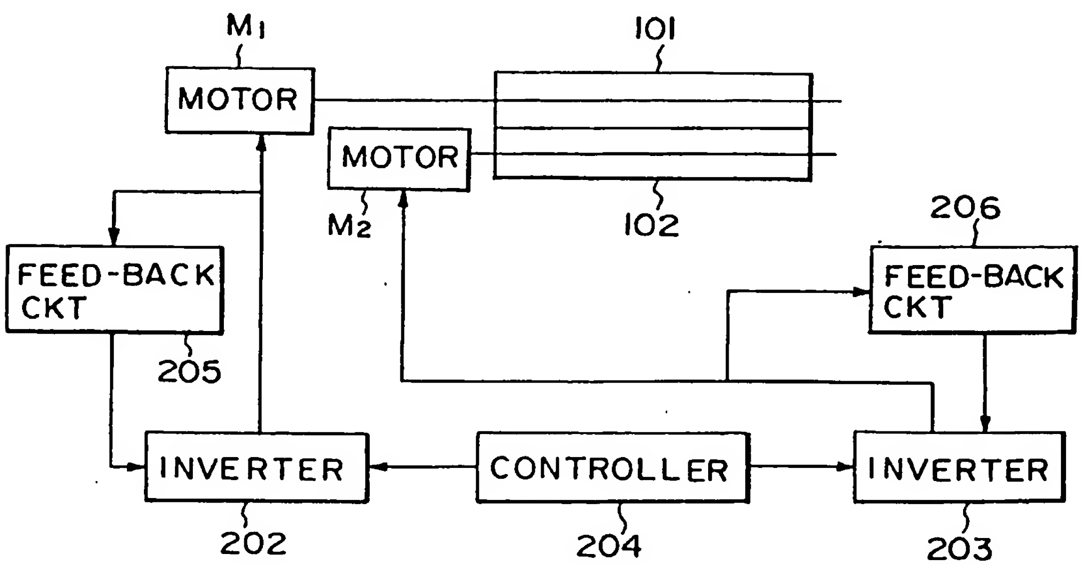
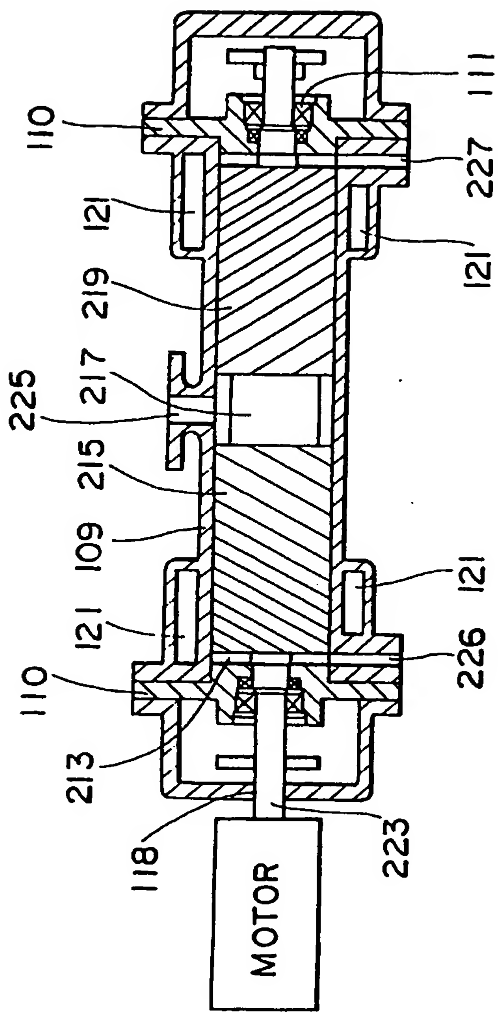

M1
101
MOTOR
MOTOR
M2
102
206
FEED-BACK CKT
205
FEED-BACK CKT
INVERTER
CONTROLLER
INVERTER
202
204
203

Fig. 19



110
111
227
121
121
219
217
225
215
109
121
121
226
110
213
118
223
MOTOR

Fig. 21

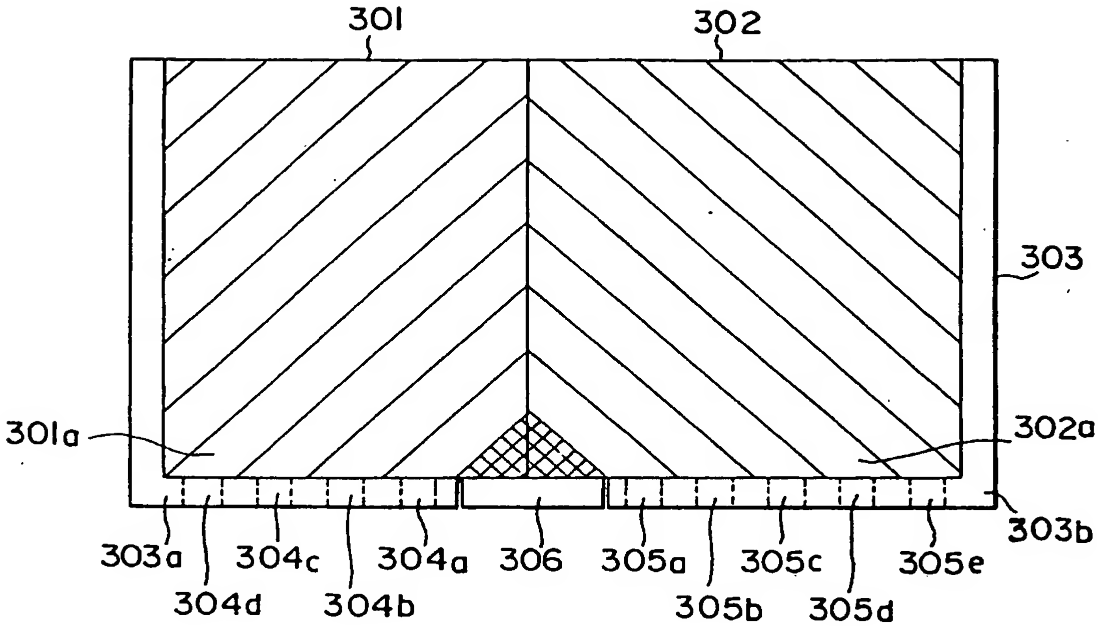
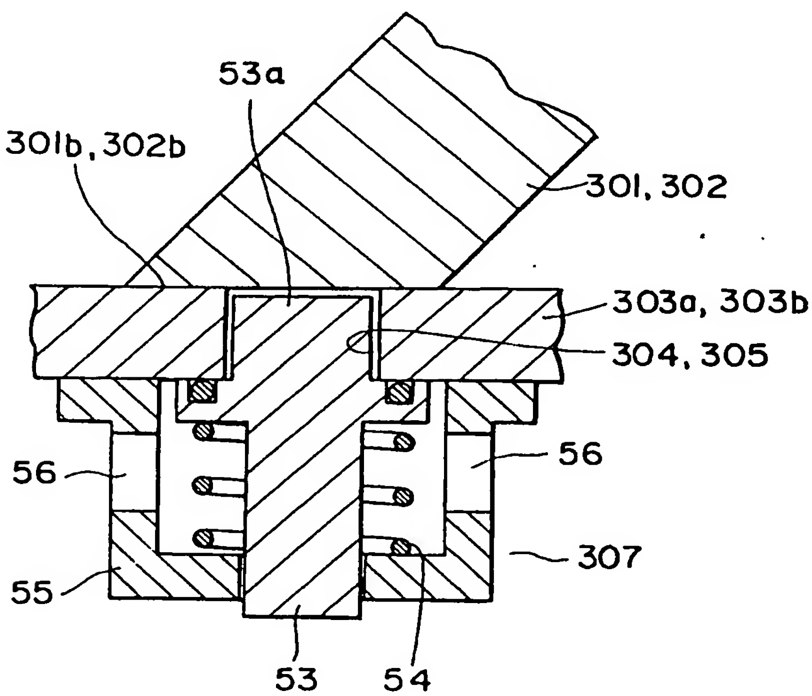

301
302
303
301a
302a
303b
303a
304c
304a
306
305a
305c
305e
304d
304b
305b
305d

Fig. 22


53a
301b, 302b
301, 302
303a, 303b
304, 305
56
56
307
55
53
54